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# TWO-PHASE FLOW OF OIL-REFRIGERANT MIXTURE THROUGH THE RADIAL CLEARANCE IN ROLLING PISTON COMPRESSORS

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**Abstract**

Gas leakage through the radial clearance is an important cause of volumetric efficiency loss in rolling piston compressors. A good understanding of its mechanism is essential in any attempt to increase the volumetric efficiency of the compressor. In this work the gas leakage is considered to be caused by a mixture composed by oil and refrigerant dissolved in it. The mixture flow is modeled as a stationary homogeneous two-phase flow. The results show that the bubbles formation due to the solubility reduction of the refrigerant in the oil changes the flow characteristics significantly, reducing the mass flow rate of both mixture and refrigerant gas, mainly at low temperatures. I was found that the mass flow rate of the refrigerant gas calculated using the two-phase flow model can be 30 % lower than the mass flow rate of the refrigerant gas estimated using the single-phase flow model.

## INTRODUCTION

The volumetric efficiency of the rolling piston compressor is related to the refrigerant leakage, clearance space, suction gas heating, return of the gas through the discharge valve and oil lubricant flow. Among them, the refrigerant leakage is the main cause of the volumetric efficiency loss. A great amount of the refrigerant leakage occurs at the existing clearance between the external surface of the rolling piston and the internal surface of the cylinder, known by radial clearance. Figure 1 shows a typical position of the radial clearance and the main components of the rolling piston compressor pump. Krueger (1988) has estimated that about 30% of total internal loss of the refrigerant gas are due to this leakage. Therefore, a good understanding of the gas leakage through the radial clearance is important in estimating and improving the volumetric efficiency of the compressor.

A review of the main works published in the last 20 years involving leakage modeling at the radial clearance shows that for most investigations simplified models have been considered. Several authors have dealt with this problem assuming compressible flow of pure refrigerant gas. Pandeya e Soedel (1978), Yanagisawa and Shimizu (1985a), Xiuling *et al.* (1992), Zhen and Zhiming (1994) e Huang (1994) are some of the representative literature on this subject. Other authors have considered the presence of the oil at the radial clearance but have not included the influence of the refrigerant gas dissolved in the lubricant; examples are Lee and Min (1988) and Leyderman and Lisle (1995). Gasche *et al.* (1997) have presented a broad discussion about several phenomena associated with the flow through the radial clearance and have proposed various models to calculate this flow, Gasche *et al.* (1998a), Gasche *et al.* (1998b) and Gasche *et al.* (1999). In all their models Gasche *et al.* assumed that the gas

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leakage through the radial clearance was caused by either pure oil flow or single-phase oil-refrigerant mixture flow.

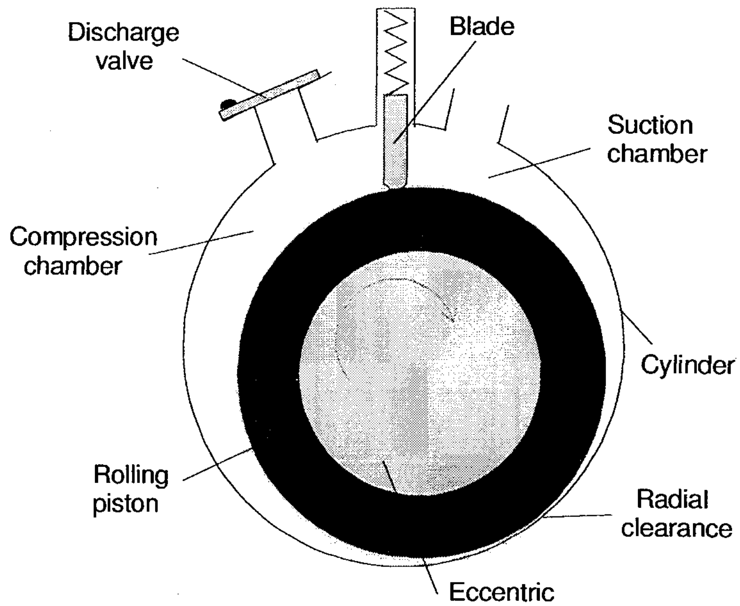


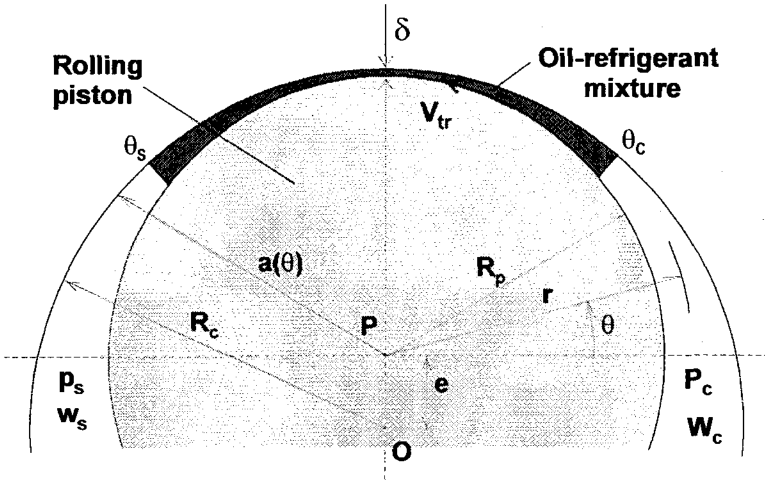

Figure 1. Schematic representation of the rolling piston compressor

Costa *et al.* (1990) performed an experimental visualization to verify the flow pattern through the radial clearance during a normal operation of the compressor. The results of this work show two important characteristics of the flow. The authors have firstly observed that part of the radial clearance is always filled with a liquid fluid. Besides, they have noted the presence of a great number of bubbles in the liquid, most of them concentrated just after the minimal clearance value. Actually, this liquid is not a pure substance but a mixture of the lubricating oil and the refrigerant gas being pumped. In fact, the refrigerant gas is dissolved in the lubricant oil, resulting in an homogeneous mixture. The amount of refrigerant gas dissolved in the oil depends on both the oil temperature and the gas pressure. As the oil-refrigerant mixture flows through the clearance, the pressure drops and the solubility of the refrigerant in the oil decreases resulting, instantaneously, in a supersaturated mixture. To reach a new equilibrium condition, part of the refrigerant dissolved in the oil converts into bubbles that break off the mixture flow at the suction chamber. This amount of gas consists of a mass loss because it must be recompressed. Accordingly, a realistic model to calculate gas leakage through the radial clearance must consider the two-phase flow of the oil-refrigerant mixture.

In the present work it is performed an investigation of the gas leakage through the radial clearance where the oil-refrigerant mixture flow has been modeled as a stationary two-phase flow using an homogeneous model.

## PROBLEM FORMULATION

The geometry used to study the oil-refrigerant mixture flow in the radial clearance is shown in figure 2, in which only part of the radial clearance is filled with the liquid mixture. This is in accordance with the visualization performed by Costa *et al.* (1990). Also shown in the figure are the dimensions required in the problem formulation. At the beginning and at the end of the liquid mixture, $\theta_c$ and $\theta_s$, respectively, surface tension effects were ignored and a flat interface was considered as shown in figure 2.




$$a(\theta) = \sqrt{e^2 \operatorname{sen}^2 \theta + R_c^2 - e^2} - e \operatorname{sen} \theta$$

Figure 2. Geometry for the flow at the radial clearance

Momentum and continuity equations govern the isothermal oil-refrigerant mixture flow, which can be modeled as two-dimensional since the channel width, characterized by the rolling piston thickness, $H_p$, is much larger than the radial clearance. Furthermore, because $\delta$ is very small fluid acceleration can be ignored and the momentum equation can be written considering only the balance between the viscous and pressure forces, as in a typical lubricating problem. Using cylindrical coordinates this results in the following differential equation,

$$\frac{\partial}{\partial r}\left(\mu r \frac{\partial u}{\partial r}\right) = \frac{dp}{d\theta} \tag{1}$$

The velocity profile at any cross section can now be determined integrating equation (1) along the radial direction. Noticing that at $r=R_p$, $u=V_{tr}$ and at $r=a(\theta)$, $u=0$, where $V_{tr}$ is the absolute tangential velocity at the external surface of the rolling piston, it is then obtained,

$$u = \frac{1}{\mu}\frac{dp}{d\theta}\left[r - a - (R_p - a)\frac{\ln(r/a)}{\ln(R_p/a)}\right] + \frac{V_{tr}}{\ln(R_p/a)}\ln(r/a) \tag{2}$$

Next, an equation for the pressure should be sought. To that extent steady state mass conservation at the radial clearance can be written as,

$$\frac{1}{r}\frac{\partial}{\partial r}(\rho r v) + \frac{1}{r}\frac{\partial}{\partial \theta}(\rho u) = 0 \tag{3}$$

Substituting the velocity profile given by equation (2) into equation (3), and integrating in the radial direction from $r=R_p$ to $r=a(\theta)$, results in,

$$\frac{d}{d\theta}\left[-\frac{\rho}{\mu}\frac{dp}{d\theta} f_1(\theta) + \rho V_{tr} f_2(\theta)\right] = 0 \tag{4}$$

where,

$$f_1(\theta)=\left(a-R_p\right)\left[\frac{a-R_p}{2}-\left(\frac{R_p-a}{\ln\left(R_p/a\right)}-R_p\right)\right] \tag{5}$$

and,

$$f_2(\theta)=\left[\frac{R_p-a}{\ln(R_p/a)}-R_p\right] \tag{6}$$

Equation (4) can now be solved yielding the pressure profile along the flow. The mass flow rate of the mixture can then be obtained using the determined pressure profile for integrating equation (2). To calculate both the pressure and the velocity distributions it is necessary to specify the geometric characteristics of the clearance, the boundary pressure at the compression and the suction chamber, $p_c$ and $p_s$, respectively, and the physical properties, $\mu$ and $\rho$. These properties are unknown variables and will be calculated considering a two-phase flow model as will be explained next.

The mixture flow through the radial clearance is taken as a two-phase flow, and the bubble formation is due to the reduction of refrigerant solubility in the oil, w. In accordance with the experimental visualization performed by Costa *et al.* (1990), the homogeneous model appears to be a good approximation to represent this flow. In the homogeneous model the two-phase flow behaves like a single-phase flow having physical properties whose values are, in some sense, mean values for the flow. Yanagisawa and Shimizu (1985b) proposed the following equations to calculate these mean values at any position along the flow,

$$\rho=\alpha\rho_g+(1-\alpha)\rho_l \tag{7}$$

$$\mu=\alpha\mu_g+(1-\alpha)\mu_l \tag{8}$$

where $\rho_g$ is the gas density and $\rho_l$ is the liquid mixture density; $\mu_g$ is the gas viscosity and $\mu_l$ is the liquid mixture viscosity. The void fraction $\alpha$ is the ratio of the area occupied by the gas to the area occupied by the liquid mixture at each particular cross section. The void fraction is obtained from the quality, x, using the following equation,

$$\alpha=\frac{1}{\left[1+(1/x-1)\,\rho_g/\rho_l\right]} \tag{9}$$

To determine the quality, x, it is considered that the liquid mixture remains always saturated with refrigerant. Therefore, the refrigerant in gaseous phase at any cross section along the longitudinal direction, $\theta$, is determined from the difference between the solubility of the refrigerant at the inlet, $w_c$, and the solubility of the refrigerant at that particular section, w. A mass balance from the inlet to a generic position along the flow yields the following equation for the local quality, x, as function of the local refrigerant solubility, w,

$$x=\frac{w_c-w}{1-w} \tag{10}$$

where $w_c$ is the refrigerant solubility in the oil at the compression chamber conditions, that is, $T_{mix}$ and $p_c$.

The refrigerant solubility, w, required in equation (10) is calculated using the local pressure, obtained from the solution of equation (4), and the mixture temperature, considered constant in the present work. The equation proposed by Sakurai and Hamilton (1984) was used to carry out this computation. The density and the viscosity of the liquid mixture, $\rho_l$ and $\mu_l$, respectively, were calculated using equations proposed by Sakurai and Hamilton (1984).

## SOLUTION METODOLOGY

The second order differential equation for pressure, equation (4), was solved using a finite volume methodology, Ferziger and Peric (1996). Staggered meshes with respect to pressure were employed for the u velocity. The algebraic equations were solved through the Tri-Diagonal Matrix Algorithm. More details on the discretization as well as on other aspects of the numerical methodology and solution can be found in Gasche (1996).

The final mesh used to generate the results to be presented here has 120 nodal points in the $\theta$ direction. Finer meshes were tested without showing greater improvement in the results. A channel length of $\pi/3$ rads was adopted to represent the length of the radial clearance that is filled with the oil-refrigerant mixture. For that, the beginning and end locations of the liquid film were taken as $\theta_c=\pi/3$ and $\theta_s=2\pi/3$, respectively. Other channel lengths were tested and no significant influence was observed. This is so because most of the pressure drop occurs very close to the minimum radial clearance as will be seen latter. At the beginning and end of the liquid film pressure remains virtually constant and an extension of the film length has no effect on the pressure profile along the flow. The compressor dimensions used are: $R_p$=20.11 mm, $R_c$=23.00 mm and $H_p$=27.00 mm, and refer to a small refrigeration compressor. All the calculations were performed for a mixture composed by refrigerant R22 and mineral oil ISO VG 46.

## RESULTS AND DISCUSSIONS

The main goal of this work is to determine the mass flow rate of refrigerant gas that leaks from the compression chamber to the suction chamber using the two-phase flow model, and to compare it with the value obtained using the single-phase flow model developed by Gasche *et al.* (1998b). The only difference between these two models is in the computation of the physical properties. In the single-phase flow model these properties are taken as constant along the flow and are calculated using a mean value for the refrigerant solubility along the clearance.

To apply the proposed stationary two-phase flow model, time mean values of both the pressure at compression chamber, $p_c$, and the tangential velocity of the rolling piston, $V_{tr}$, are used. These mean values are obtained integrating their instantaneous values during a single compression cycle. Gasche (1996) determined the instantaneous values of the compression pressure and the tangential velocity modeling the compression process and the rolling piston dynamics, respectively. These time integration resulted in $p_c$=1.41 MPa and $V_{tr}$=-0.0266 m/s. The refrigerant solubility at the inlet of the flow, $w_c$, was determined using the mean value of the compression pressure, $p_c$, and the temperature of the mixture, $T_{mix}$.

Figure 3 shows the longitudinal profiles of the main variables of the flow for a typical case, in which $\delta$=10 $\mu$m and $T_{mix}$=100 $^0$C. From the figure it is seen that as the liquid mixture approaches the minimum clearance ($\theta=\pi/2$) the pressure drops abruptly reducing the refrigerant solubility in the oil. In turn gas is released from the liquid increasing both the mixture void fraction and kinematic viscosity.

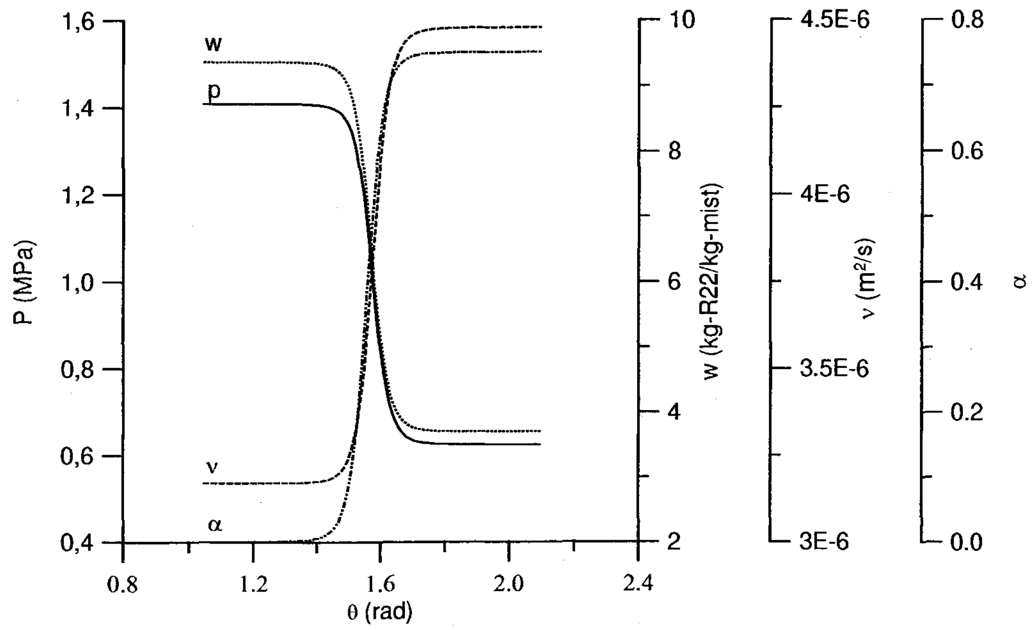



Figure 3. Values of the main instantaneous variables along the flow for $\delta=10$ μm and $T_{mix}=100\ ^0C$

Table 1 presents mass flow rates results for both the mixture and the refrigerant in gaseous phase. The gaseous mass flow rate represents the refrigerant mass loss through the radial clearance, and it is computed from a refrigerant mass balance at the flow exit resulting in,

$$\dot{m}_{gas} = \dot{m}_{mix} \frac{w_C - w_S}{1 - w_S} \tag{11}$$

where $w_s$ is the refrigerant solubility at the flow exit, obtained from the pressure at the suction chamber, $p_s$, and the mixture temperature, $T_{mix}$. The mixture mass flow rate, $\dot{m}_{mix}$, is calculated integrating the velocity profile, given by the equation (2). Because the mixture mass flow rate is constant throughout the clearance, the integration of the velocity profile can be performed at any circumferential location.

All mass flow rates calculated using the two-phase flow model are lower than those calculated using the single-phase flow model. The main reason for that is the higher values of the mixture kinematic viscosities computed from the two-phase model. Furthermore, the reduction in the mass flow rate between the models decreases as the temperature increases because of the reduction in the refrigerant solubility at the film inlet. This reduction yields a lower raise in the kinematic viscosity. For example, for $T_{mix}=80\ ^0C$ the kinematic viscosity increases about 80% whereas for $T_{mix}=120\ ^0C$ the kinematic viscosity increases just about 20%.

The results obtained with the two-phase flow model proposed in this work indicate that it is very important to consider refrigerant outgasing in evaluating refrigerant loss through the radial clearance.

Table 1. Mixture mass flow rate and rate of refrigerant leakage for various cases investigated

| | $T_{mix}$=80 °C | | | $w_c$ =0.1482 kg-R22/kg-mix | | |
|---|---|---|---|---|---|---|
| | $\dot{m}_{mix}$ (g/s) | | | $\dot{m}_{R22}$ (g/s) | | |
| δ (μm) | Single-phase | Two-phase | $\Delta\dot{m}$ (%) | Single-phase | Two-phase | $\Delta\dot{m}$ (%) |
| 10 | 0.269 | 0.183 | 32.0 | 0.0262 | 0.0178 | 32.0 |
| 20 | 1.535 | 1.042 | 32.1 | 0.1494 | 0.1014 | 32.1 |
| 40 | 8.692 | 5.890 | 32.2 | 0.8460 | 0.5733 | 32.2 |
| 60 | 23.92 | 16.21 | 32.2 | 2.329 | 1.577 | 32.3 |
| 80 | 47.07 | 33.24 | 32.3 | 4.777 | 3.235 | 32.3 |
| 100 | 85.72 | 58.05 | 32.3 | 8.343 | 5.650 | 32.3 |
| | $T_{mix}$=100 °C | | | $w_c$ =0.0936 kg-R22/kg-mix | | |
| 10 | 0.272 | 0.219 | 19.5 | 0.0160 | 0.0129 | 19.4 |
| 20 | 1.553 | 1.247 | 19.7 | 0.0913 | 0.0733 | 19.7 |
| 40 | 8.789 | 7.050 | 19.8 | 0.5167 | 0.4145 | 19.8 |
| 60 | 24.19 | 19.40 | 19.8 | 1.422 | 1.140 | 19.8 |
| 80 | 49.62 | 39.78 | 19.8 | 2.917 | 2.339 | 19.8 |
| 100 | 86.67 | 69.48 | 19.8 | 5.095 | 4.085 | 19.8 |
| | $T_{mix}$=120 °C | | | $w_c$ =0.0470 kg-R22/kg-mix | | |
| 10 | 0.2959 | 0.2603 | 9.0 | 0.00821 | 0.00748 | 8.9 |
| 20 | 1.630 | 1.480 | 9.2 | 0.0468 | 0.0425 | 9.2 |
| 40 | 9.221 | 8.367 | 9.3 | 0.2648 | 0.2403 | 9.3 |
| 60 | 25.38 | 23.02 | 9.3 | 0.7288 | 0.6611 | 9.3 |
| 80 | 52.06 | 47.22 | 9.3 | 1.495 | 1.356 | 9.3 |
| 100 | 90.92 | 82.47 | 9.3 | 2.611 | 2.368 | 9.3 |

## CONCLUSION

Steady two-phase flow of oil-refrigerant mixture through radial clearance in rolling piston compressors has been modeled to estimate leakage of refrigerant gas. An homogeneous model has been used to represent refrigerant outgasing and the influence of the gaseous phase is included in both density and absolute viscosity, through apparent physical properties. The results obtained with the two-phase flow model show that the bubbles formation changes significantly the flow characteristics, reducing the mass flow rate of both mixture and refrigerant gas, mainly at low temperatures. It has been found that the mass flow rate of the refrigerant gas calculated using two-phase flow model can be 30 % lower than the mass flow rate of the refrigerant gas estimated using single-phase flow model.

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